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FLOW BEHAVIOUR IN CENTRIFUGAL IMPELLERS

By

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Abstract

Detailed investigations through hot-wire anemometry at the outlet of a centrifugal compressor impeller reveal a complex nature of flow behaviour both spatially across the blade to blade pitch and with respect to time. The relative flow angle and velocity do not fit into the conventional steady state jet-wake model nor the one-dimensional slip theory. A hypothesis that periodic vortices shed from the suction surface well inside the impeller sweeping across the impeller passage is put forward to explain the large variations in flow angle observed.

Nomenclature

- d diameter
- m mass flow rate
- P pressure surface
- S suction surface
- T time for one pitch
- U impeller tip speed
- θ angular location of particle from the suction surface
- ρ density
- ϕ flow coefficient, $\dfrac{4m}{\rho_{01} d_2^{\,2} U_2}$
- λ mass flow fraction through wake
- ε ratio of wake width to blade pitch

Subscripts

- $-$ average
- r radial
- t tip
- o total
- 1 impeller inlet
- 2 impeller outlet

1.Introduction

The flow in centrifugal impellers is highly complex in nature influenced by the effects of rotation, viscosity and turbulence. The blade to blade flow field due to rotation is required to balance coriolis force in the relative channel flow. This would induce secondary flows in the hub and shroud boundary layers. The flow is further influenced by the secondary flow generated by the velocity or pressure gradients in the inlet section of the inducer (1,2) and leakage flow between the blades and the stationary casing. The turbulence could stabilize or destabilize the flow depending on the impeller channel curvature. Flow destabilization on the pressure surface hub portion causes low velocity particles to accumulate on the suction surface shroud (3). This region is called the wake (4).

Quite often the impeller outlet flow is assumed to be steady with property variations across the blade to blade as well as hub to shroud channel. Quasi three-dimensional potential flow theory (5,6) or empirical modelling such as jet-wake flow are then used to analyse the flow within the channel and to compute outlet flow conditions (7,8,9). The impeller outlet flow models are reinforced by measuring the actual flow using instruments like laser velocimeter and hot-wire anemometer (10, 11). This paper describes the measurement and analysis of centrifugal impeller outlet flow using hot-wire anemometry system. The measurement of the impeller outlet flow in the present investigation indicate relative velocity and relative flow angle do not fit into the conventional jet-wake model nor the one-dimensional slip theory (12). A hypothesis that periodic vorticies shed from the suction surface well inside the impeller sweeping across the impeller passage is put forward to explain the large variations in flow angle observed.

2.Test Facility

A centrifugal impeller of 525 mm diameter 45.5 mm width with 23 vanes backswept by 40 degrees with reference to radial direction was rotated at 5000 rpm by a D.C motor. Thyristor control with feed back for the D.C motor ensured maintenance of the speed to an accuracy of 0.1%. An electronic torquemeter coupled in between the gear box and the compressor was used to measure the speed and input power. A bell mouth in the inlet duct was used to ensure uniform flow to the compressor. A throttle plate at the exit of the volute casing was used to vary the mass flow rate through the impeller.

3.Instrumentation

The test facility was well instrumented for detailed flow measure-

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ments at impeller outlet. A hot-wire anemometer placed 8 mm radially outwards of the impeller as shown in Fig. 1 was used in two angular positions to measure radial and whirl components of absolute velocity in a dynamic mode across the blade to blade pitch at the mid channel between hub to shroud width of the impeller at outlet. A linearizer was used in conjunction with the hot-wire anemometry circuit and calibration was carried out separately in a steady uniform flow before and after the experiments. The hot-wire traces were

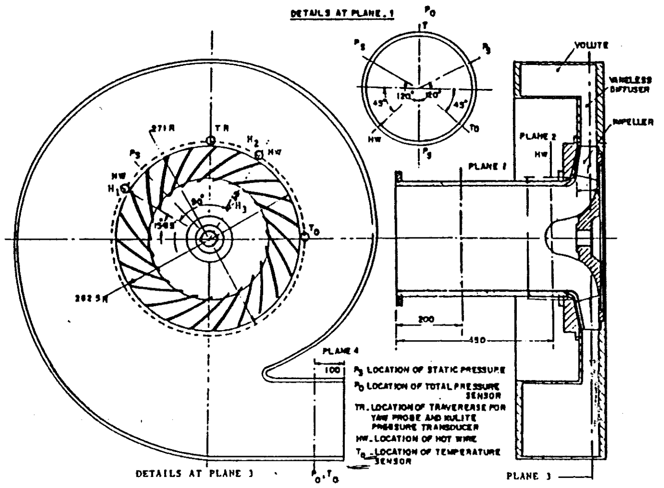


Fig. 1 Test impeller instrumentation

captured through a computer controlled dual beam signal analyser (FFT Analyser) and recorded through its memory on to magnetic discs. The signal analyser is connected to the computer through an HP interface bus. An once per revolution spike generated from a magnetic pickup and shaft projection was used to trigger the hot-wire trace and record the same for a duration, of 50 milli-seconds, corresponding to nearly 4 revolutions of the impeller. Fifty such recordings one after the other in the phase locked manner was obtained to get the ensemble average of signal across 92 blade passages. The blade passing frequency is about 1.92 KHz. Additionally a combination probe with yaw holes and a central pitot tube was used to measure average absolute velocity and absolute flow angle, which would corroborate the hot-wire measurements. Mass flow through the impeller was measured through a traverse in the inlet duct. The input power and speed were measured using an electronic torquemeter from which no-load bearing losses and estimated disc friction losses were deducted to compute the fluid dynamic power imparted to the fluid. The time averaged flow measurements were carried out through an on-line data acquisition system. Power and mass flow measurements were accurate to the order of $\pm$ 4%.

4. Results and Discussions

Hot-wire traces of radial component of absolute velocity for two blade passages near the hub, mean and shroud of impeller width for three different flow coefficients are shown in Fig. 2. At

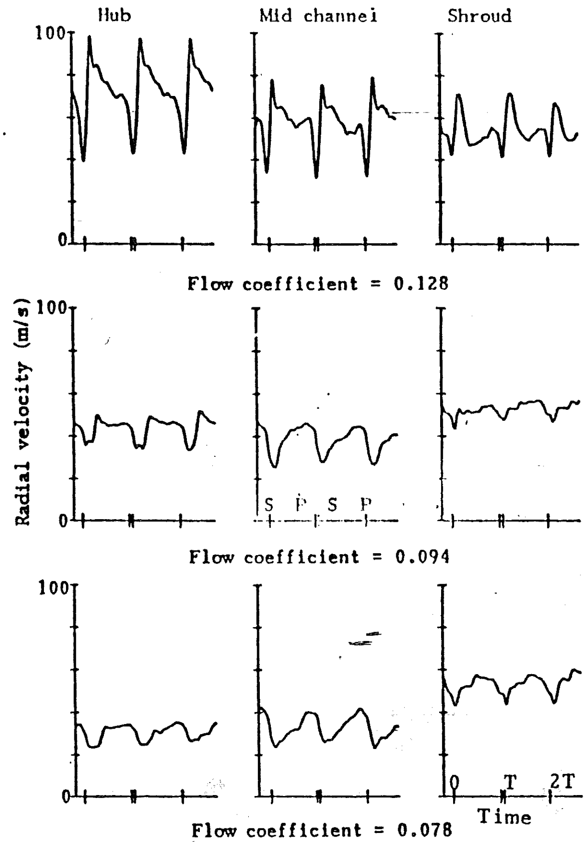


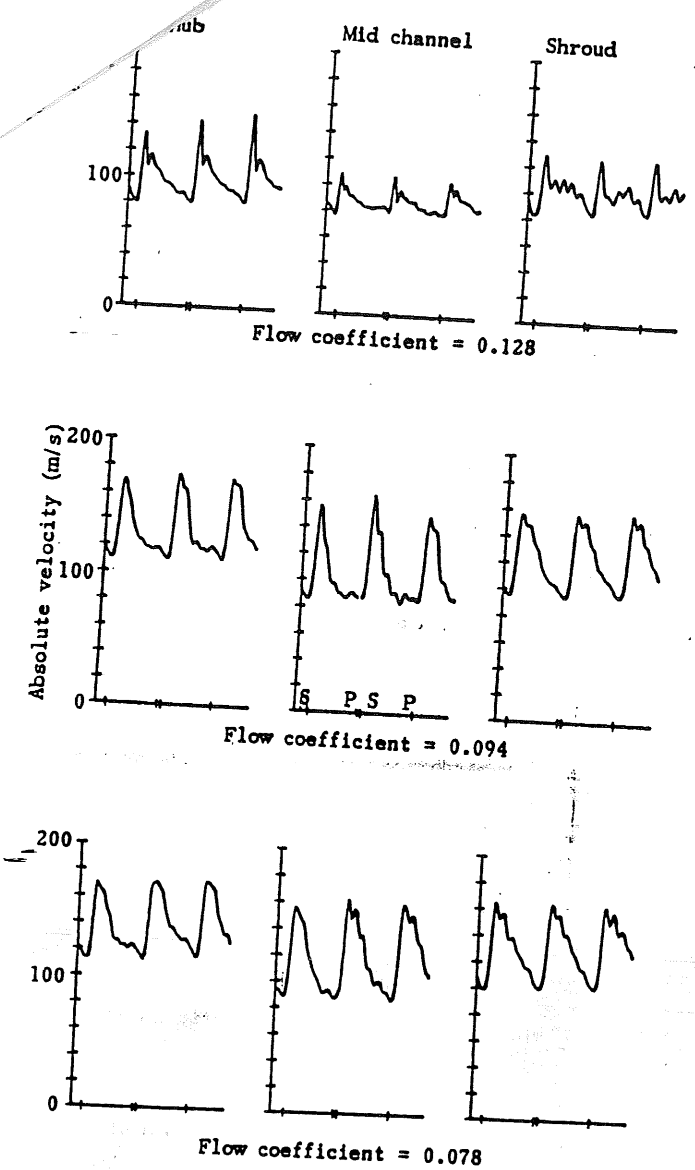
Fig. 2 Ensemble averaged instantaneous radial velocity variation at impeller outlet.

lower flow coefficients the total mass flow rate at impeller outlet is shared between the two regions with a lower value in the suction half of the blade passage and a higher value in the rest of the passage. The integrated mass flow from hot-wire measurements of radial velocity agrees well with the mass flow measurement at inlet and with that computed from average total pressure measurements. The comparison is given in Table-I.

Table 1

Estimated mass flow rate

Flow coefficient	0.128	0.094	0.078
From three hole yaw probe	4.30	3.16	2.78
From hot-wire anemometer	4.51	3.35	2.86
From calibrated wall static pressure	4.60	3.27	2.83



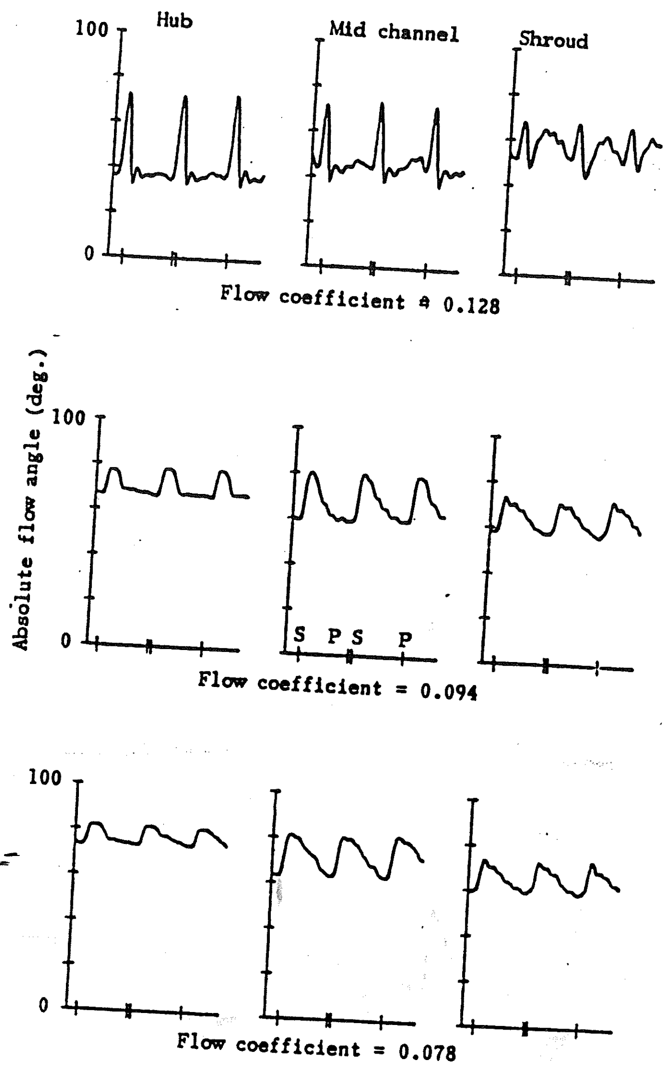


Fig. 3 Ensemble averaged instantaneous absolute velocity variation at impeller outlet.

Fig. 4 Ensemble averaged instantaneous absolute flownangle variation at impeller outlet.

Absolute velocity and absolute flow angle at impeller outlet as observed from hot-wire measurements are given in Fig. 3 and Fig. 4 . It is seen that the good flow uniformity which exists at the mean width of blade passage at high flow coefficients, gets deteriorated as the flow coefficient is reduced and the loading is increased through higher incidence of flow. A region of high absolute velocity flowing at large angles is clearly seen to be growing by the side of the suction surface within the passage. It is to be noted that the radial component of absolute velocity in this region and hence the mass flow fraction here is much lower than the rest of the passage. A comparison of these values viz. mass flow fraction in the wake, wake width ratio, wake to jet velocity ratio are given in Table-II. At the optimum flow coefficient, the mass flow fraction and the relative wake width are close to the values reported in literature (7,10) and generally used in a steady state jet-wake model for design analysis of centrifugal impellers.

Table II

Jet-wake flow parameters

Flow coefficient	0.094	0.078
Mass flow fraction, λ	0.205	0.316
Wake width ratio, ϵ	0.250	0.400
Relative velocity ratio, ν	0.400	0.420
Radial velocity ratio	0.790	0.730

Local instantaneous values of absolute whirl component and the flow angle are very much larger in the suction surface (wake) region as compared to elsewhere in the passage. This behaviour is exhibited near mean width position at lower flow coefficients. Near the shroud section, such a behaviour, though exists at all flows, is very much clear at low flow coefficients. Measurements in the present investigation indicate much larger values for local work input. These are dynamic measurements portraying instantaneous values.

Analysis of flow structure in terms of velocity and angle particularly in the relative frame would provide more clarity. Using the measurements of radial and tangential components of flow and impeller peripheral speed, the relative flow velocity and flow angle could be computed. This is shown in Fig. 5 with full lines at the mid channel. The flow computed using jet-wake model is also shown in this figure by chain lines. In this model the flow at impeller outlet is considered to be separated into two regions, a jet region near the pressure surface wherein potential flow as per design exists and a wake region near the suction side having a reduced relative velocity, with a shear layer separating the two regions. The relative flow angles in the two regions, however are assumed to be nearly same as the value estimated from slip and blade outlet angle to substantiate the presence of shear layer. This would give rise to a variation of slip in the respective regions. The reduced relative velocity in the wake region leads to a larger absolute velocity and reduced slip, hence a slip factor of unity with zero slip is generaly taken for this wake region. This implies that the relative flow angle in the wake will be equal to the blade angle and in the jet it is higher than the blade angle by an amount slip. The estimated relative velocity from the measurements shows that a distinct jump across the passage with reduced value near suction surface and increasing to a higher value near pressure surface. However, the relative flow angle, instead of being nearly the same as the blade angle a large variation is seen in the wake region near the suction surface.

Hot-wire measurements were collected for nearly 200 revolutions, with the 23 blade passages traversing the probe for each revolution. Considering the flow in every channel to be identical and periodic, the time-wise measurements across the passage can be taken as flow property distribution in any passage at that instant time of measurement. From this data, the absolute velocity variations at a specified angular location across the blade to blade passage with respect to time can be

Full line - From hot-w measurements.
Chain line - From conceptua model.

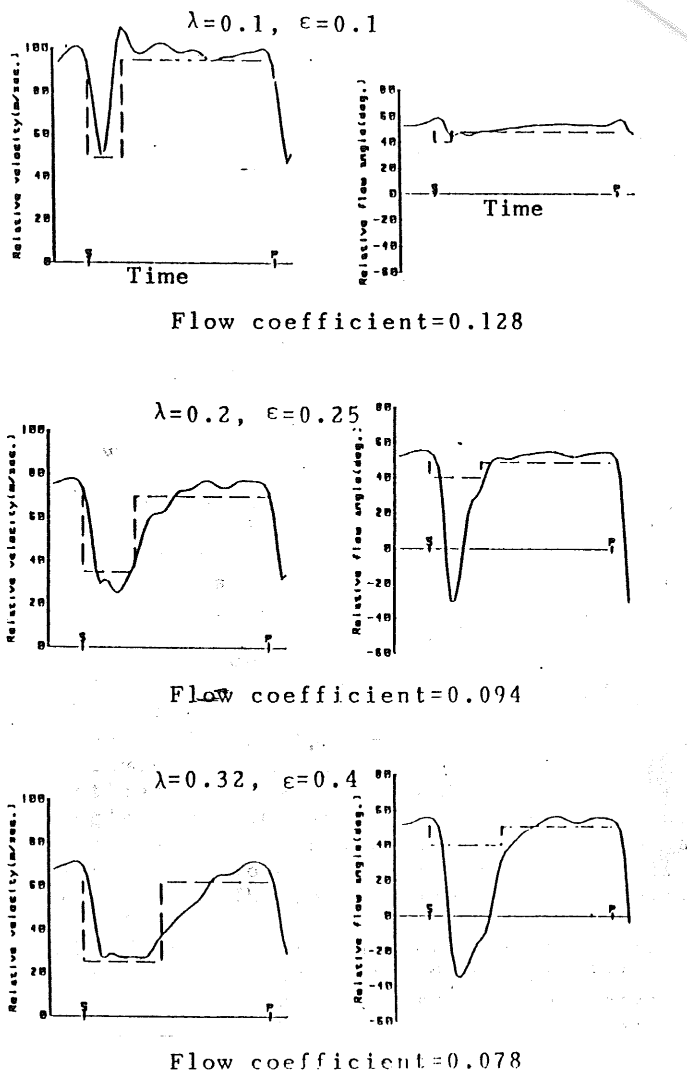


Fig. 5 Comparision of relative velocity and relative flow angle as measured and estimated from conceptual jet-wake model.

obtained at different flow coefficients and these are shown in Figs. 6.a to 6.c corresponding to mid channel. The angular location from the suction surface of the forward blade, covering the passage and the average value of velocity is given along side of each plot at the right. Two blades indicating suction and pressure surfaces are also marked in this figure. The abscissa is the time, normalised with reference to time for duration of measurement namely four revolutions. The ordinate is marked with velocity variation with reference to the average value. Markings to indicate the time duration for each revolution as well as each blade passage are also shown.

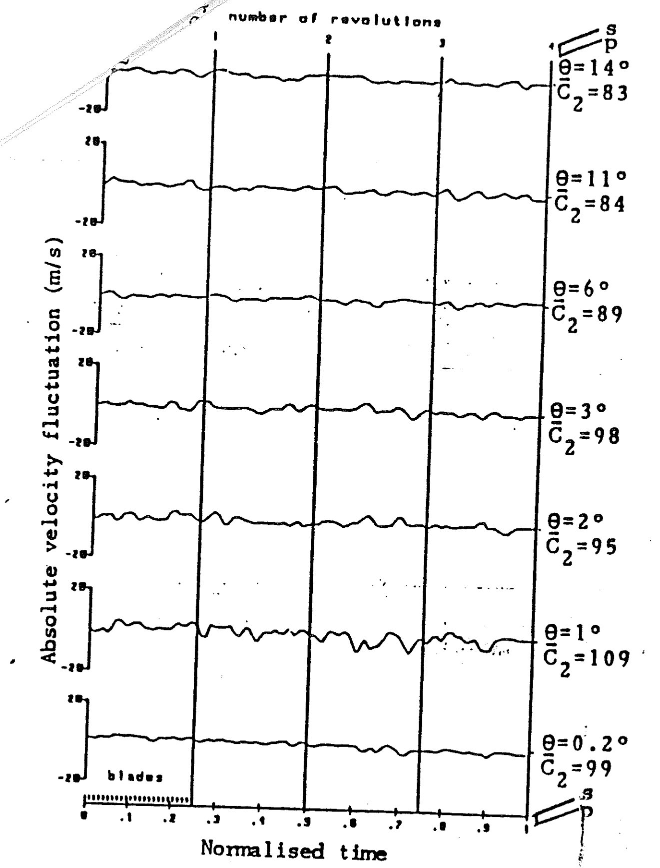


Fig. 6a Unsteady fluctuations in absolute velocity at flow coefficient = 0.128

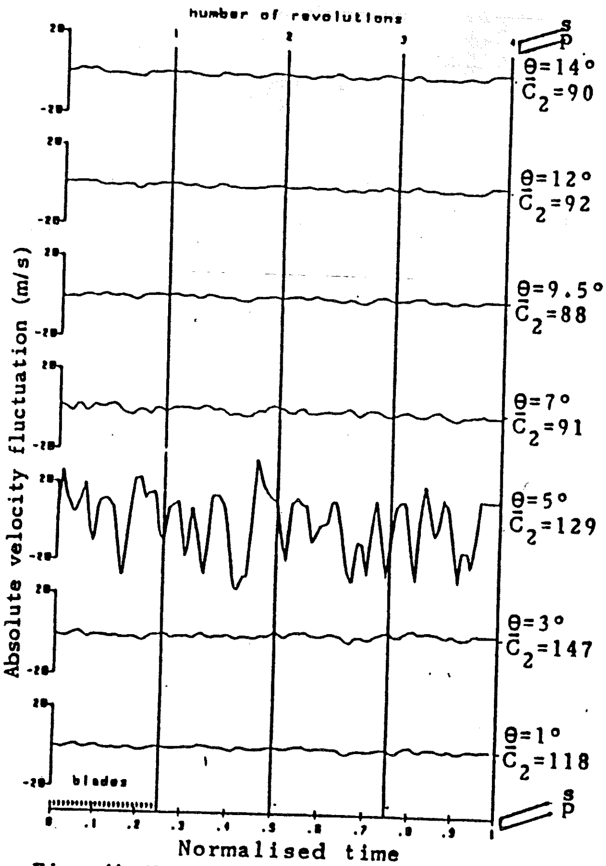


Fig. 6b Unsteady fluctuations in absolute velocity at flow coefficient = 0.094

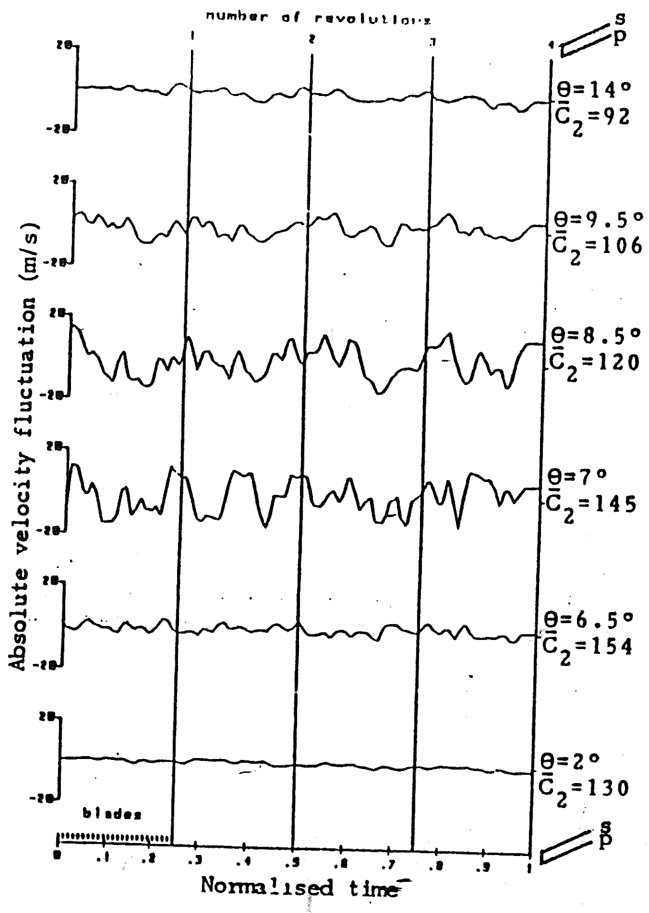


Fig. 6c Unsteady fluctuations in absolute velocity at flow coefficient = 0.078

At the highest flow coefficient, 0.128 (Fig. 6.a) the fluctuations in absolute velocity are small in magnitude and they occur very close to the suction surface of the blade at an angular location of around, θ=1 deg. As the flow coefficient is reduced to 0.094 the fluctuations in absolute velocity increases in magnitude and moves away from the suction surface (Fig. 6.b). At this flow coefficient large fluctuations in absolute velocity are noticed at an angular location of θ=5 deg. from the suction surface. With further reduction in flow coefficient to the lowest value, the large fluctuation in velocity occurs over an extent of angular location from 7 to 9 deg. (Fig. 6.c). The location of large time unsteady variation of flow depends on the value of flow coefficient. The unsteady variation of flow moves away from the suction surface as the flow coefficient is reduced. Such large variations of flow velocity much more than average value at different flow coefficients are characteristic of a swirling or vortex flow. It is to be noted that this manifestation has been observed in the absence of rotating stall or any drop in impeller pressure rise. In fact, the compressor exhibited an optimum efficiency at the mean flow coefficient of 0.094. The time unsteady variations in flow are rather large enough to deviate from the picture of flow behaviour one gets from a jet wake model based on a steady separation from a point on the suction surface. A reasoning that low energy particles of a turbulent flow tending to move and accumulate near the suction surface shroud based on Richardson number values (3) would not satisfactorily explain the large values of absolute velocity and its variation with time in an orderly manner as observed.

The variation of absolute velocity both across the blade to blade passage near the suction surface region and with respect to time could find an explanation from a probable hypothesis that periodic shedding of vortices, from the suction surface, sweep across the passage. For instance, vortices near the suction surface could be assumed to be formed and shed into the flow from inside the impeller. The flow is clean and undisturbed in between two instances of vortex formation. Such vortex shedding and attached flow would be cause and effect of each other and alternating at periodic intervals. This vorticity having been originated due to the rotational vorticity of the impeller can be expected to move opposite to the direction of rotation in the relative frame and sweep across the passage. At the same time, it would indeed be pushed radially outwards and be diffused by the flow. Such a hypothesis as above would provide logical explanations for the time-wise and localised variations of whirl velocity and increase in flow angle. As a result there is a large increase in specific power input in this region, more than what would result through a decrement in mass flow in the region. As the vortices get diffused within a short range and move outside of the impeller, the flow near the pressure surface is unaffected by this and is rather steady.

The flow coming out of the impeller entering into the diffuser is spatially non-uniform at a given instant of time and unsteady in nature. The stabilised uniform flow coming alongside the pressure surface is chased and penetrated by the unsteady flow with a large whirl component coming from the suction surface (wake) region of the succeeding passage. The mixing of the two would be much more rapid than a wake mixing process behind a stationary blade would suggest or for that matter behind an axial rotor. During this mixing process behind the outlet of the impeller the flow for considerable time is at a larger angle than the average absolute angle computed from conservation of angu[illegible] The diffusion within the init[illegible] of a vaned diffuser would be in[illegible] by this large variation in flow ang[illegible]

5. Conclusions

Measurements using hot-wire anemometry indicated that

* There is a large variation in velocity and flow angle across blade to blade pitch.

* Good flow uniformity which exsists at the mean width of the blade passage at high flow coefficient gets deteriorated as the flow coefficient is reduced.

* The calculated relative flow velocity and in particulat relative flow angle do not fit well into conventional steady state jet-wake model nor one-dimensional slip theory.

* There is a large unsteady variation of flow over a short region between blade to blade. The location of unsteady variation of flow moves away from the suction surface as the flow coefficient is reduced.

* The unsteady variation of flow can be explained by a hypothesis of periodic vortex formation inside the flow channel and sweeping across the passage as it comes out of the impeller.

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